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STUDY OF A MAGNETIC REFRIGERATOR FOR SPACE APPLICATIONS FOR TEMPERATURES UNDER 20°K

(Note SBT/CT/86-18)

P. Seyfert

Translation of "Etude d'un Refrigerateur Magnetique a Vocation Spatiale pour des Temperatures Inferieures a 20 Kelvins", Commissariat a l'Energie Atomique Grenoble (France). Centre d'Etudes Nucleaires. Final report. May 26, 1986. CENG-Note-SBT/CT/86-18. ESA-CR(P)-2311. ETN-87-99866. pp.1-48

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16. Abstract

A thermodynamical analysis of the integration of a magnetic cycle in a refrigeration cycle; the process configuration choices; and the implementation of the first design of a last stage refrigeration step within a minimum overall weight design concept are summarized. The feasibility of magnetic cooling to 4K design temperature is demonstrated. Compared to Joule-Thompson systems, overall weight is of the same order.

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STUDY OF A MAGNETIC REFRIGERATOR FOR SPACE APPLICATIONS FOR TEMPERATURES UNDER 20° K

(Note SBT/CT/86-18)

by P. Seyfert

Cryogenic Laboratory

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ON CONTRACT EXECUTION

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SUMMARY

The objective of the contract is the study of a refrigerator for space applications which, by incorporating a magnetic cycle, can reach liquid helium temperatures. The study pushed as far as possible the limits of current knowledge of magnetic refrigeration.

The work done includes a thermodynamic analysis of how to conceptually integrate a magnetic cycle into a refrigeration system, process configuration choices, a rough draft of a last stage refrigeration step, and a comparative analysis of the weights of these systems.

The study demonstrates the feasibility of magnetic refrigeration to approximately 4^{o} K of objects sent on long term missions. This conclusion, however, is dependent on experimentally demonstrating the functionning of the magnetic cycle selected. When comparing the magnetic approach with the traditional Joule-Thomson (J-T) approach, it appears that both systems weigh approximately the same when installed in a space craft. Both approaches require the same type of mechanical refrigeration for pre-cooling.

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1. INTRODUCTION

Installation on space craft of equipment needing a cryogenic environment is currently envisioned for missions lasting up to 5 years. As one examines p.1* techniques available for low power refrigeration units which are compatible with the rigors of such missions, the most critical specification is that they must operate without being serviced or maintained for several years. This is even more critical than their ability to operate in a semi-weightless environment.

Obviously, the degree of difficulty increases as the required temperature decreases. An evaluation study of refrigeration concepts applicable to space missions was conducted in 1984 by teams from four laboratories, including the author's, for the European Center for Space Research and Technology (contract ESTEC-SNIAS No. 5629/N?PB). The results of this study indicated that in the area of temperatures around 4° K--which is of particular interest to us-- it is convenient to make a distinction between the problem of pre-cooling from 300° K to 15-20° K and the problem of the last stage which must allow reaching temperatures below 10° K.

For only the pre-cooling, gas cycles appeared to afford technically feasible solutions. Stirling and Villeumier cycles seemed to be the best candidates. The development of "space versions" requires the perfection of guide and drive systems of cold and hot moving parts (pistons, gas displacers) which have neither friction nor connecting rod assemblies thus eliminating all sources of wear.

As for the last stage, there is only one type of gas cycle, J-T, which has the technical possibilty of attaining useful power outputs of approximately 0.1 Watt at 4° K with a reasonable yield. A very large development effort is to be expected if the reliability level of this technology must reach the standards required for space missions.

In this situation, it is indispensoble that the possibilities offered by new techniques be carefully studied. Among potential alternatives to gas cycles, magnetic refrigeration, is certainly one of the more interesting. Current knowledge suggests that this technology is particularly adaptable to temperatures below 20° K. Research programs concentrating on the temperature range between 15 and 4° K are underway in several laboratories throughout the world--including

*Numbers in margin indicate -1- foreign pagination.

the author's.

The goal of the present study is to complete, based on information currently available, a rough draft of a design for a magnetic refrigerator functionning as the last stage of a refrigeration system for space applications. Because, as mentioned above, the J-T cycle is the only alternative among existing technologies it served as a reference point every time comparisons were possible.

The work proceeded following the "work schedule" defined contractually. p. 5
It had been initially anticipated that the rough draft would lead to a development and testing plan. During the final phase of the study, however, it turned out that fundamental research on magnetic refrigeration had not progressed as far as had been hoped and it was premature to prepare precise technical specifications for a prototype. For this reason it was agreed that the contractor would study variants at early stages of the rough draft as well as monitoring new developments on magnetic materials.

The final report outlines in a condensed form all of the results obtained during the study. For this it drew on intermediate technical reports and progress reports.

2. THERMODYNAMIC ANALYSIS OF CONCEPTS p. 6

2.1 General

The contribution which magnetic refrigeration can make to the general problem of refrigeration at temperatures below 20° K is based on two distinct concepts: (a) a magnetic refrigeration stage assumes the refrigeration function at the cold source (for example, 4.5° K) or (b) magnetic refrigeration is used to "assist" the last stage of a gas cycle refrigerator in order to increase its refrigeration capacity. Magnetic refrigeration thus does not intervene directly at the cold source.

The overall thermodynamic efficiency of a refrigerator is defined by the ratio of the extracted refrigeration power, Q_E and the total mechanical power furnished, W; i.e., Q_E/W. We will use the inverse of this expression which we will call specific consumption.

$$W/Q_E = \text{Specific consumption}$$

It turns out that in the two refrigeration concepts considered here the major part (>90%) of the mechanical energy comes from the pre-cooling done at approximately 15-20° K. If one calls T_C the temperature at this level the overall specific consumption is thus practically equal to:

$$\frac{W}{Q_E} = \left[\frac{W}{Q(T_C)}\right]_P \cdot \frac{Q(T_C)}{Q_E} \qquad (1)$$

The first term of this expression represents the specific consumption of the pre-cooling auxiliary refrigerator. The second term, which one can call specific thermal removal, is the ratio of the thermal energy removed at T_C and the refrigeration energy. We wil first consider the two terms separately and combine them later. p. 7

2.2 Specific thermal removal of the cold stage

The first of the two concepts mentioned above consists of utilizing the demagnetization of an active material as the exclusive refrigeration source in the cold stage. The fluid present in the refrigerator (helium) is the heat exchange medium and is practically isobaric. The model adopted for this concept is very simple: the refrigerator is described by a Carnot cycle reduced by the actual efficiency η . The values of η are between 0.6 and 0.8. If the temperature at the cold source is T_E and T_C is the temperature at the hot source, the specific thermal removal at temperature T_C is given by the following equation:

$$\frac{Q(T_C)}{Q_e} = \frac{T_C}{\eta\, T_E} \qquad 0.6 \leq \eta \leq 0.8 \qquad (2)$$

The simplicity of this model is explained by the lack of useful data. There are still too few experiments in this area to be able to introduce more concrete models. In fact, it appears to us preferable to call equation (2) a realistic working hypothesis.

In the second concept envisioned, refrigeration at the cold source is caused by a gas cycle modified by a magnetic cycle. Keeping in mind the intended application and the current state of the art of gas cycles, the J-T cycle can be envisioned for the present case. The role of the magnetic refrigeration consists of cooling a certain fraction of the principal flow in the high pressure branch of the cycle up-stream of the cold exchanger (Fig. 1). p. 8

Leaving the magnetic stage, the flow undergoes an iso-enthalpic expansion and is re-injected at the cold extremity of the exchanger in the low pressure branch of the cycle. The aim of this approach is to compensate the dis-equilibrium of the cold exchanger caused by the increase of the specific heat of helium at high pressures. In high energy refrigerators this "doping" of the cold exchanger is habitually caused by a turbine or a piston engine which, because of their design, are absolutely useless for the weak flows considered here.

Fig. 1a shows the last stage of a J-T cycle modified to include our concepts. The enthalpy diagram of helium (Fig. 1b) allows one to follow the thermodynamic transformations of the gas as it passes through different parts of the cycle. From this cycle scheme, we have calculated the total specific thermal removal which is now the sum of two contributions, one coming from the pre-cooling loop of the principal cycle and the other coming from the appended magnetic stage.

p. 9

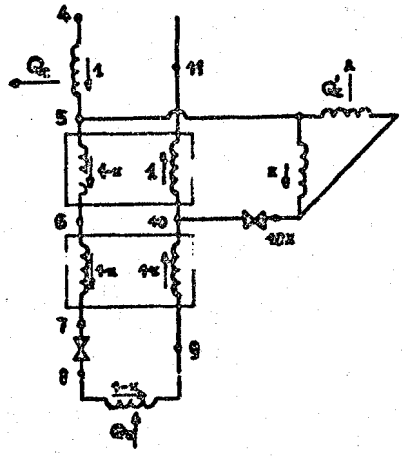

Figure 1a Joule-Thomson cycle "doped" by magnetic refrigeration.

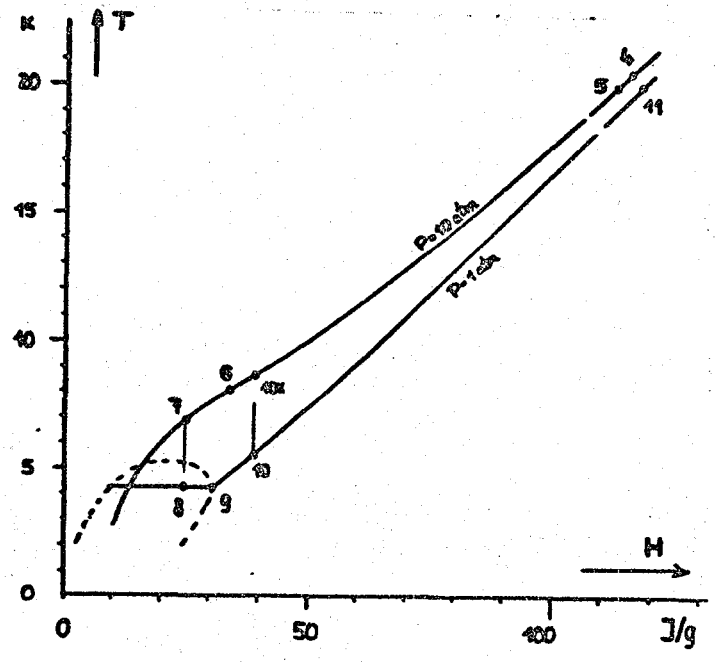

Figure 1b Temperature-enthalpy diagram of helium.

A working hypothesis of the thermodynamic model of the magnetic stage was established--the reversible cooling of gas flow mx at temperature T_c involves heat removal. p. 10

$$\dot{m}x\ T_c\ \Delta S$$

where ΔS is the entropy difference of helium between points 5 and 10 x in Fig. 1a. In assuming, as before, an actual efficiency of η heat removal of the magnetic stage becomes:

$$Q_c' = \dot{m}x \frac{T_c \Delta S}{\eta} \qquad 0.6 \leqslant \eta \leqslant 0.8 \qquad (3)$$

To calculate heat removal, T_c, in the pre-cooling loop of the principal cycle (between points 4 and 5 of Fig. 1) we constructed a numerical model based on the actual thermodynamic properties of helium and the following plausible assumptions:

-Difference in temperature between the high pressure and low pressure branches on the warm side of the exchangers: $T = 0.02T$.

-Temperature of the first pre-cooler : 80° K

-Low pressure value : 1 atm.

-Temperature at point 10 (Fig. 1) : 5.5° K

The value of the fraction of the flow diverted to the magnetic stage [$\dot{m}_x$ of equation (3) shown as "subsidiary flow" in Fig. 2] was chosen in such a way as to bring the cycle to the boundaries of violating the "2nd Law". The thermodynamic yield y is maximal.

p. 11

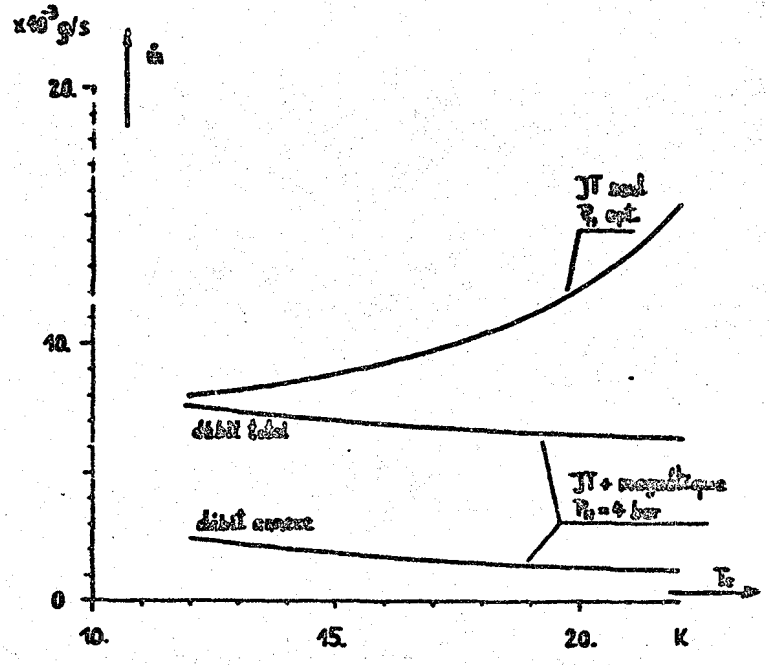

Figure 2*

Helium flow in the J-T cycle as a function of pre-cooling temperature.
Cold energy: 0.1W at 4.2° K

The major result emanating from a comparison between the thermodynamic efficiency thus calculated and those of a classical J-T cycle is that a mixed system allows operation at very low values at high pressures while maintaining efficiencies close to optimal values.

Figure 3 plots the thermal energy removed at the heat source as a function of its temperature for a useful energy of 0.1W at 4.2° K. Three cases are illustrated: refrigeration using only a magnetic stage [according to equation (2)], refrigeration using a J-T stage operating at 4 atm. and "doped" by p. 12

*débit total = total flow
débit annexe = subsidiary flow
JT seul = only Joule-Thomson

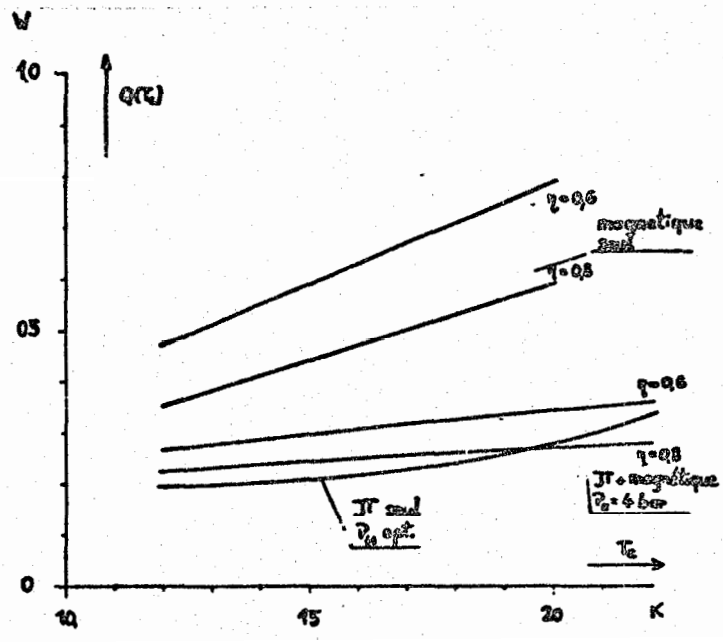

Figure 3*

Heat removal of the cycles studied as a function of the pre-cooling temperaturee.

Cold energy: 0.1W at 4.2° K.

magnetic refrigeration [$Q'_C + Q_C$, equation (3) and numerical model described above] and finally refrigeration using a classical J-T stage operating at optimum high pressure values (13.5 atm. for T_C = 12° K, 18.5 atm. for 15° K, and 25 atm. for 20° K; numerical model described above with $\dot{m}x = 0$).

One sees that for a given temperature of the heat source and up to approximately 20° K, the classical J-T cycle has the poorest heat removal capability. It is clearly followed by the mixed cycle operating at 4 atm. Refrigeration using only a magnetic stage affords the highest heat removal capacity. p. 13

Before drawing conclusions on this point it is appropriate to examine the actual efficiency of existing refrigerators which will be used for pre-cooling at the heat source.

* - - - - - - - - -

magnetique seul = only magnetic (cooling)
JT seal = only Joule-Thomson

2.3 Actual efficiency of refrigerators utilizable for pre-cooling

It is evident from Fig. 3 that for removal temperatures between 10 and 20^{o} K the needed pre-cooling capacity is in the range of 0.2 to 1W.

We have analyzed the results of a recent review of the state of the art (worldwide) of small cryogenic refrigerators (J.L. Smith, Jr., G.Y. Robinson, Y. Iwasw, MIT, Dept. of Mechanical Engineering, Cryogenic Engineering Laboratory; January 1984). Existing refrigerators applicable to the temperature range of interest to us operate solely using "breeder" gas cycles. In the majority of cases it consists of a Gifford-MacMahon cycle or a variation thereof--a modified Solvay cycle. If one plots the specific consumption of these machines against their useful power one obtains a surprising result (Fig. 4): in the power range of 0.2-2W the best performers are those situated along a line which, when plotted logarithmically, appears as a straight line with a slope of -45^{o}.

p. 14

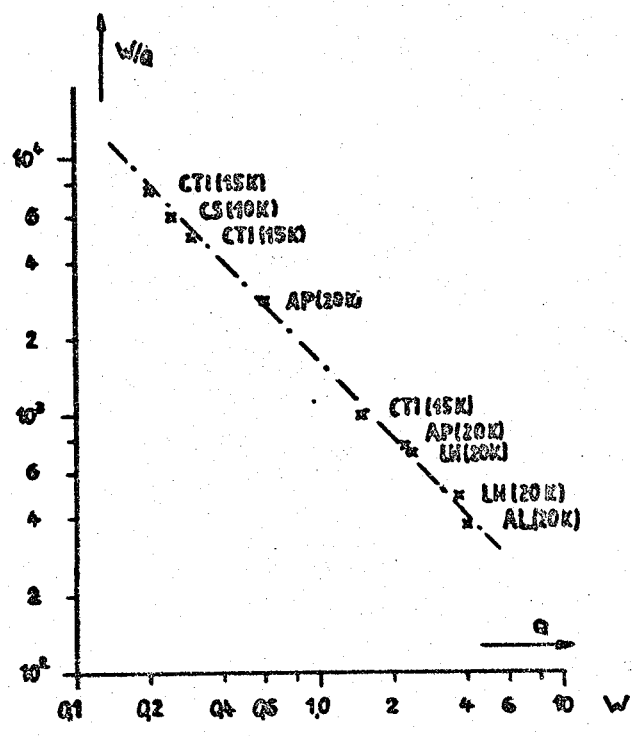

Figure 4

Specific consumption of current, low power mechanical refrigerators in the temperature interval 10-20° K.

Initials of manufacturers: AL, L'Air Liquids; AP, Air Products; CS, Cryosystems; CTI, CTI-Cryogenics; LH, Leybold-Heraeus.

This indicates that the specific consumption is inversely proportional to the operating power or that the absolute value of the consumption is constant (ca. 1,700W) regardless of the cold operating poorer or the temperature (in the range of 10-20° K). These results can be interpreted to indicate the impossibility of reflecting the miniturization of these machines in their consumption.

There remains only the machines envisioned for pre-cooling refrigerators for space applications which are primarily Stirling or Villeumier types rather than Gifford-MacMahon types (see Introduction). Unfortunately, in the 10-20° K temperature range, up to now industrial experience does not exist with these

cycles. As they are part of the same family of "breed" cycles, there is no evidence that they will perform any differently than current machines.

2.4 Conclusions from the thermodynamic analysis

If one agrees with the results of the preceeding paragraph, the three types of stages analyzed in paragraph 2.1 consume approximately the same amount of energy. In effect, the variations in the level of heat removal between 10 and 20° K (see Fig. 2) has no effect on this consumption and thus ceases to be a factor in making a choice.

It follows that other factors must be considered for the selection including the most appropriate design of the magnetic refrigeration stage as well as the most favorable temperature level for heat removal. Several of these elements, notably the overall weight of the refrigerator and associated auxilliary systems, will be studied in the chapter devoted to the rough draft.

3. PROCESS CONFIGURATION CHOICE p. 16

3.1 General

The two concepts of magnetic refrigerators described in the preceeding chapter assume magnetic cycles capable of maintaining temperature variations on the order of 10° K of a cold source at approximately 5° K. Given the properties of magnetic materials, the use of magnetic Carnot cycles will lead to magnetic fields on the order of 6 to 7T. These induction levels are considered to be incompatible with the rigors of industrial--let alone space--applications. Consequently, the use of magnetic cycles with internal exchangers would permit a significant reduction in the field strength required.

Using magnetic cycles with an internal exchanger is currently being studied in several laboratories including the author's. The first efforts must lead to research on the most appropriate configuration. In the remainder of this chapter, we will present three possible configurations and we will attempt to determine their relative advantages and disadvantages.

3.2 Rotating machine

Figure 5 schematically shows the configuration of a rotating refrigerator.

The active magnetic material is in a ring completely bathed *in* an atmosphere of gaseous helium. A highly perforated structure affords a large surface area

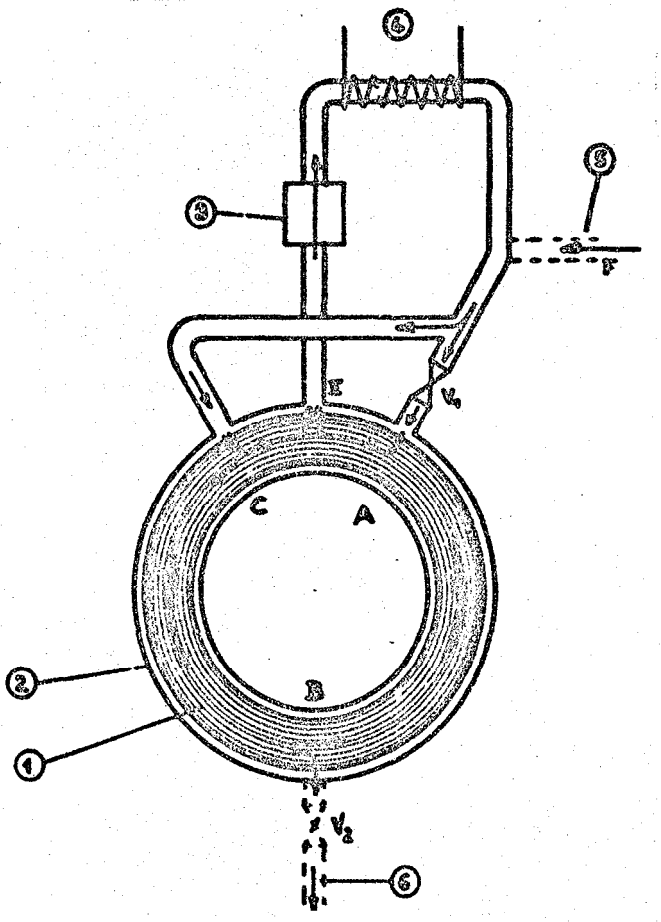

Figure 5

Schematic configuration of a rotating machine.

1. Active material
2. Annular surrounding wall
3. Helium circulator
4. Pre-cooling loop
5. External helium feed
6. Exit

⟹ Direction of material rotation

Direction of helium circulation

for gas contact and thus assures efficient heat exchange. The ring turns freely inside an immobile annular surrounding wall whose function is to contain the helium gas and to thermally isolate the assembly from the environment. p. 18

A circular magnetic system was omitted from the scheme in Fig. 5 for the sake of clarity. This system imposes a well defined field profile on the material. It is zero at C and rises rapidly towards A where it is at its maximum. Along the perimeter A - B - C, it follows a carefully adjusted form which progressively returns it to zero at C.

A helium circuit containing several taps through the annular surrounding wall allows control of gas circulation in different parts of the apparatus. Because of a circulator up-stream of a pre-cooling loop, a large flow of helium whose temperature is well defined penetrates at points C and A of the wall. The majority of the gas takes the shortest path towards exit E while passing by the section of the ring finding itself at each instant at that place. Throttle valve V1 causes a certain depression at A relative to C. A small fraction of the flow entering at C is thus directed to the longer route (following the arrows in our scheme) and travels from C - B - A before being collected at E.

At point F of the helium circuit it is possible to introduce a flow of helium from outside. It can be recovered at point B of the apparatus through a tap on the wall at this point and by appropriate regulation of valve V2. In the case of a mixed refrigerator, flow coming from the J-T cycle would take this route. p. 19

In the case of a self-contained magnetic refrigerator the entering and exit taps, as well as valve V2, would be omitted. The cold source is situated in the zone around point B.

It is easy to convince oneself that at the end of one complete circuit of the ring, each portion of the magnetic material will have travelled the same thermodynamic cycle covering the entire temperature interval between the cold and hot sources. This is a peculiarity of rotating machines which can prove to be awkward as it reduces refrigeration possibilities (temperature range for example) to the limits characteristic of a single material.

From the point of view of thermodynamic functions this configuration appears well defined. There are, however, some mechanical functions in a toroidal

geometry which are particularly difficult to be confident of.

First of all there is the problem of driving the ring. How can the ring be made to turn without perturbing the movement of helium across the different parts of the torus (which will inevitably lead to a deterioration of its operation)? As of now, we know of no adequate solution applicable to our project.

In addition, there are problems associated with the mechanical resistance of the body of the active material. The ring whose structure is, by necessity, highly perforated could be constructed only with difficulty to precise dimensions with a satisfactory stiffness. To be realistic, one must thus count on the appearance of some torsion in the ring to which is added a rather large uniaxial pull caused by azimuthal variations of field and temperature levels. p. 2

To counter these forces and to minimize the resulting deformations, it is reasonable to expect that a guidance system along the entire perimeter of the ring--such that the ring will be centered in the interior of the toroidal wall--will be needed. One can imagine one or more guides. Incorporating these guides will inevitably introduce mechnical play between the mobile ring and the annular surrounding wall which risks creating shortened paths such that warm gas will reach the cold part of the apparatus thus seriously compromising its operation.

Plausible assumptions on "breeder" helium flow and losses of charge in the perforated ring structure and shortened paths leads us to estimate that a section of 0.5 to 1.0 mm^2 suffices to cause a thermal by-pass leading to a drop in actual efficiency of 20% for the magnetic stage. It is not certain, however, that the complexity of the toroidal geometry allows the construction of an apparatus with much less mechanical play.

In summary, one can say that major difficulties are apparent in finding technical solutions to both mechanical problems and technical problems related to the rotating configuration. p. 21

3.3 Linear machine with active material moving in alternate directions

In comparison with the rotating configuration, machines with a linear geometry offer a priori a better chance of achieving a compromise between the

constraints due to mechanical problems and those arising from thermal problems.

We have explored in more detail two possibilities: the first one considered is characterized by pistons of mobile active material. Figure 6 schematically shows the principal elements of this approach.

Two pistons, provided with active material, move inside fixed cylindrical walls which are thermally isolated from their environment. The pistons undergo very slow axial oscillations controlled by external means not shown in Figure 6 (bearings, linear drive).

Each cylindrical wall is placed in the air-gap of a magnetic system which creates a well defined field the length of the travel path of the two pistons. Magnetic induction reaches its maximum value at the high point (relative to Figure 6) of its path.

p. 2

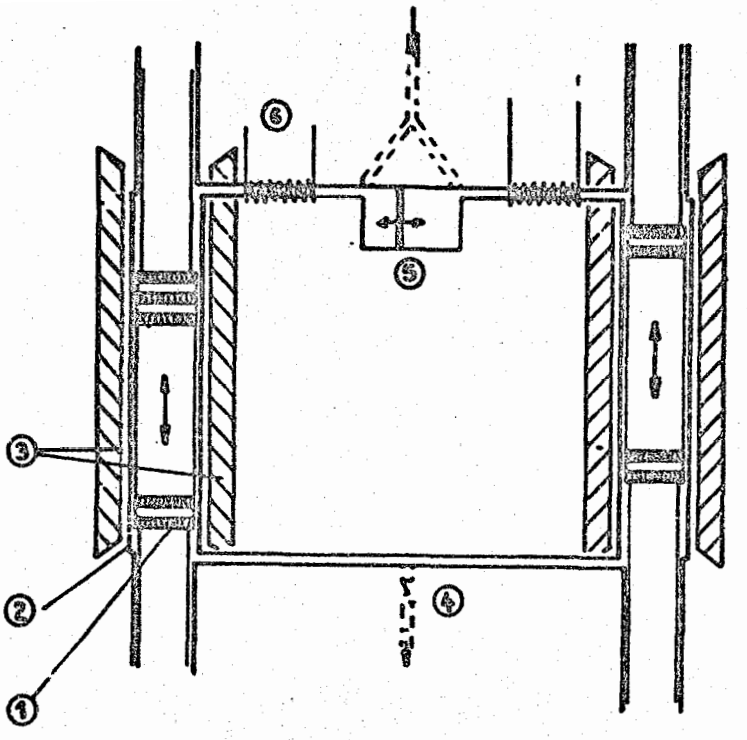

Figure 6

Schematic configuration of the linear machine with active material moving in alternate directions.

1. Piston of active material
2. Cylindrical wall
3. Magnetic system
4. Bleed valve
5. Piston displacer
6. Pre-cooling loop

The highly perforated structure of the pistons allows facile passage p. 23
of an axial flow of a gaseous flux of helium giving it a large heat exchange surface. A tube ties together the "lower" extremities of the two cylindrical walls. Two other tubes are tapped into the "upper" extremities. They pass by pre-cooling loops to a piston displacer chamber which causes an alternate flow of helium across different parts of the apparatus.

In the case of a mixed refrigerator in which the magnetic stage has the role of cooling a helium flow a supplementary tap equipped with a valve (4) is needed.

While functionning, the direction of the axial temperature gradient of the two pistons is invariable: the hot end is at the "upper" extremity and the cold end is at the "lower" extremity. Each section of the piston periodically executes a cycle around an average temperature which is more or less shifted with respect to the temperature of cycles followed by neighboring sections.

The alternating movement of the principal pistons, the helium one controlled by the piston displacer and, eventually, the opening of valve 4 are correlated in such a way as to assure that the four parts of the magnetic cycle occur in succession.

The technology of the linear machine affords advantages including:

p. 24

-The cylindrical form of the pistons of the active material lend themselves to rigid construction to precise dimensions.

-Integration of the magnetic pistons in their cylindrical walls and the transmission of drive and guiding forces is much easier than solving the difficulties encountered in the toroidal geometry.

The alternating movement of the magnetic pistons is the origin of an effect which leads to the transfer of stray heat from the heat source to the cold source. This is referred to as the "shuttle" effect. We thought that it would be appropriate to determine the consequences of this effect on our project.

Based on a very simplified theory of the "shuttle" effect (Zimmerman, F.F., Longsworth, R.C., Shuttle Heat Transfer, Adv. Cryog. Eng., 16, 342-351, 1971) and with the plausible assumption that a 5 cm. long piston moves 2 cm. inside a cylindrical wall of 1 cm^2 cross-section and allowing for 0.1 mm. of play between the piston and the cylinder, calculations indicate a thermal loss of 0.25W. This number is close to the 0.1W of cold operating power envisioned for our refrigerator.

Even if the calculation of this value is not totally accurate, it shows that the shuttle effect is capable of seriously hurting the eventual development of the refrigerator.

Without going further in these discussions, we have drawn the conclusion p. 25
that this type of linear machine is not of great interest for our needs.

3.4 Linear machine with stationary active material and a rotating magnetic system

The scheme in Fig. 7 shows a configuration which, although it very much resembles Fig. 6, differs from it in several essential points. The active material is enclosed in two cylindrical containers which are in a fixed position thermally isolated from their environment.

Magnetization and demagnetization of the material occurs through a turning magnetic system. The axis of rotation is parallel to the axis of the material containers. The profile of the axial field in the air-gap is adapted to the necessities of the thermodynamic cycle followed by the different layers of the material. In addition, the profile of the azimuthal field can be modeled in such a manner as to render the evolution of magnetic induction as seen by the material during an increasing field to be different than during a decreasing field. The exterior ring of the system is either fixed or mobile as a function of the requirements of the specified field.

The other elements of this configuration are familiar as they are identical to those in the preceeding machine.

p. 26

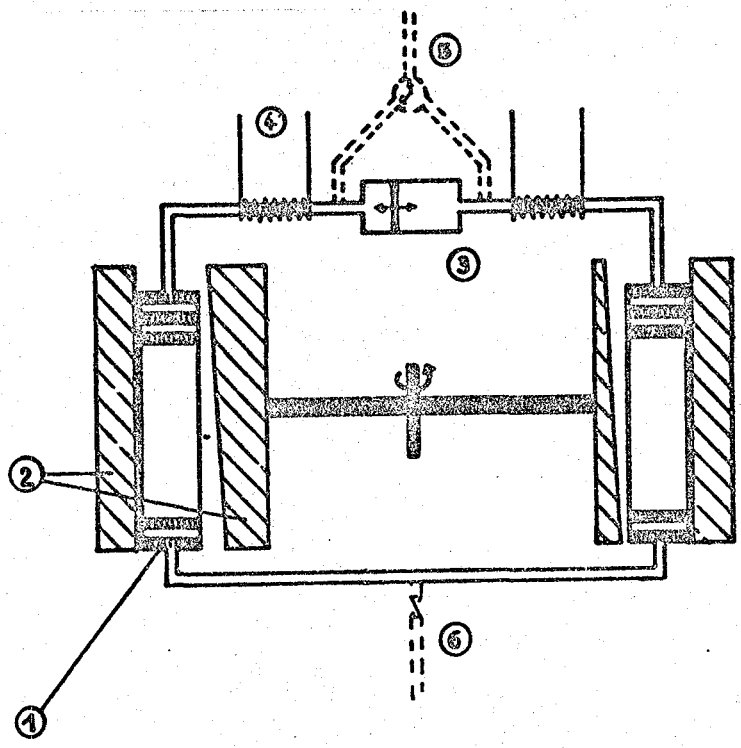

Figure 7

Schematic configuration of the linear machine with stationary active material and a turning magnet.

1. Active material container
2. Turning magnetic system
3. Piston displacer
4. Pre-cooling
5. Distribution valve
6. Bleed valve

Placing the magnetic material inside the cylindrical container allows facile axial passage of a flow of gaseous helium while affording it a very large heat exchange surface. A piston displacer is placed between the upper extremities of the containers—each of which has an exchanger serving to pre-cool the gas. A tube connects the lower extremities of the containers together. p. 27

If the magnetic stage has to cool a flow of gas a tap, equipped with a bleed valve, is installed in the lower tube. In addition, a distribution valve allows an external flow of helium to be directed to one container and then to the other.

The operating principals are completely analogous to those envisioned for the machine described in the preceeding paragraph.

With the rythym of the rotating magnetic system and the alternating circulation of helium, different layers of material in the containers go through thermodynamic cycles slightly displaced in temperature the one from the other. The overall variation in temperature which occurs between the two extremities of a column in general exceeds the range of temperatures in individual cycles. One can call this operation a "cascade cycle". In this particular case in which the cycles include refrigeration phases, it is referred to in the literature under the name "active regenerator".

Mechanical problems with this machine are much simpler than in rotating machines because the magnetic material is stationary. Except for the piston displacer and the bleed valve, there are no moving parts in the cryogenic section of the apparatus.

The problem of moving parts was, in efect, transfered to the magnetic system. Its rotation should not cause major problems principally because the exterior ring can remain stationary. p. 28

The magnetic system can be heavy and its rather open geometry can cause difficulties in attaining elevated fields. The situation is less clear concerning the thermodynamic functions of the machine.

We could not, at present, detect any causes for thermal losses. The absence of movement of the material column allows, in principle, the elimination

of all risk of thermal short-circuits or "shuttle" effects. It is true, however, that a properly functionning "active breeder"--like all magnetic breeder cycles--has yet to be demonstrated experimentally.

3.5 Justification of the choice

In the three preceeding sections we have tried to assemble appropriate data on various possible magnetic refrigerators. The project which served as a benchmark in this research is classed, by its specifications, in the low power category. p. 29

Two types of configuration, rotating and linear, were considered--the latter with two variations.

In the rotating machine, all the magnetic material successively travelled the same thermodynamic cycle between warm and cold sources. The technology of this machine proved to be very difficult.

For linear configurations, relatively simple technical solutions appeared to be available. Because of the low power of the specific project studied here, we discarded a "magnetic piston" approach. A variation with a turning magnetic field and a stationary magnetic material seems, on the other hand, to be perfectly suitable.

Currently there are no experimental results indicating the feasibility of magnetic "breeding" cycles. To the author's knowledge, the rotating configuration has only been chosen for use in Carnot cycles. As for the linear configuration, two experimental programs, one of which is in the author's laboratory, are in progress to study the "active breeder" process. Results on this subject will soon be available.

Taking all of this into consideration, a choice was made in the present study to select an "active breeder" process and then to come up with a rough draft.

4. ROUGH DRAFT p. 30

4.1 General

The rough draft of the magnetic refrigerator presented here is based on the "active breeder" principle (see paragraphs 3.4 and 3.5). In addition, two options for the design of integrating the magnetic cycle in the complete refrigeration system, described in Chapter 2, were kept. It should be re-

called that it consisted of a single magnetic stage and a mixed stage of a J-T cycle with an associated magnetic cycle.

It is understood that this rough draft constitutes a first approach whose goal is essentially to inventory the main components required and to point out their current state of development.

A complete plan of the most complex system assumed was prepared, i.e., the mixed stage (see plan CEA-CENG-SBT 1985-14 and document SBT/LCT/85-289/PS/ML). From the point of view of construction, the single magnetic stage is without any doubt the simplest (see Figure 7).

4.7 Principal secondary components

The following list enumerates the different components outside of the magnetic system and the active material and delineates in each case the degree to which the chosen technology has been developed.

Joule-Thomson exchangers are of the tubular type. Technology for them exists and is well proven coming notably from the author's laboratory. p. 31

Cold valves and the circulator. The distribution valve which serves to introduce the flow derived from the J-T cycle into the magnetic cycle operates with a turning plug. This causes very small losses of the charge. The main expansion valve of the J-T loop and the expansion valve of the magnetic refrigeration circuit are needle valves whose impedence is controlled pneumatically. The clearance of the needles is very small (several μm) necessitating the protection of these valves from plugging using adequate filters (not designed). A piston displacer was chosen for the circulator. These are cryogenic components currently under development for non-space applications.

Drive and bearing devices. The turning core of the magnetic system is driven by a motor outside the evacuated chamber (ambient temperature, atmospheric pressure). The forces on the core are taken up by two cold magnetic bearings (80^{o} K). The distribution valve mentioned above is placed at the end of the shaft so it can be driven by a magnetic coupling. The back and forth movement of the piston displacer is accomplished using a cold linear motor and a dry bearing (sintered alumina). The author suggests that these are plausible

technical solutions but current data indicates that additional development is needed.

Compressor of the Joule-Thomson circuit. For a flow of 150 l/h (at STP) p. 32 with a compression rate of 4 from 1 bar necessitates a specific development program which takes into consideration the 3-5 year lifetime required for operation on board a spacecraft.

In the above list, several items are only necessary in the mixed stage assumption. These include the J-T exchanger, the distribution valve, the two plug valves, and the compressor. On the other hand, the gas circulator, the drive train (except the distribution valve's) and the bearings are items common to both options.

4.3 Magnetic system and operation of the active material

The magnetic system and the columns of the active material are tied together and can not be disassociated in their design and sizing.

The chosen approach proposes filling the annular air-gap with a bundle of juxtaposed columns filled with active material. The columns are put in sub-assemblies of four elements situated at 90^{o} from each other. Each sub-assembly is integrated into the circuit such that the alternating circulation of gaseous helium occurs in rhythm with the rotating magnetic field.

The turning core is supposed to be the generating element of the magnetic field. Two possibilities were envisioned: using permanent magnets or a superconducting winding. A stationary exterior ring in a ferromagnetic material assures the p. 33 return of magnetic flux and, at the same time, fills the role of magnetic shielding.

Calculations of the maximum induction necessary as a function of the pre-cooling temperature were made. In all cases a refrigeration power of 0.1W at 4.5^{o} K and a rotation frequency of 1 Hz of the magnetic system were assumed.

Outside the pre-cooling temperature the number of double pairs of active material columns was allowed to vary. The active material section at the hot extremity of a column was 0.5 cm^2 in size, each sub-assembly of four elements thus included two pairs of columns with a cumulative cross-section of 1 cm^2/pair.

On the other hand, the column section varies linearly with height: at the top, at the hot extremity, the section is square (0.7 x 0.7 cm^2); at the foot, at the cold extremity, the section is rectangular (0.7 x 2.1 cm^2)-- the larger dimension being in the radial direction. In all cases the column length is 15 cm.

The active material taken as a reference is gadolinium and gallium garnet (GGG) in the form of small balls whose diameter is on the order of a millimeter. This material is available as monocrystals whose size is several hundred cubic centimeters. Another material, dysprosium and aluminum garnet (DAG) was recently noted as it has better magnetic properties than GGG above 12° K. This material, p.34 however, is anisotropic. To take advantage of its properties, the material must be oriented with respect to the direction of the magnetic field. In addition, methods for its manufactur are much less developed than those for GGG. As an experiment we have in several cases studied columns with mixtures of GGG an DAG.

Figures 8 and 9 show the results for the case of a magnetic refrigerator associated with a J-T stage and for a magnetic refrigerator alone. One sees that in the second case, with all parameters staying unchanged, the maximum field level is 0.4-0.6T higher. The influence of the pre-cooling temperature is strong: between 18° K and 12° K the field strength generally drops approximately 1T. Finally, it clearly appears that in augmenting the number of column sub-assemblies (i.e., the total cross-section of the material) one could decrease the magnetic field. This effect is more reasonable as the number of columns is small.

As these results came to light, studies of the pre-measurement of the magnetic system were undertaken. As indicated above, two types of approaches were considered: permanent magnets and superconducting windings.

For permanent magnets the use of neodymium-iron-boron (Nd-Fe-B) was envisioned. With a remanence B_r = 1.2T and a density d = 7.4g/cm^3 this material is a real improvement over samarium-cobalt ($SmCo_5$; B_r = 0.85T, d = 8.3g/cm^3). To the author's knowledge, industrial production on more or less a large scale p.35 exists in Japan, the US, and significantly Europe (Vacuum-schmelze and Thyssen in West Germany and Philips in the Netherlands). The assumptions made for the

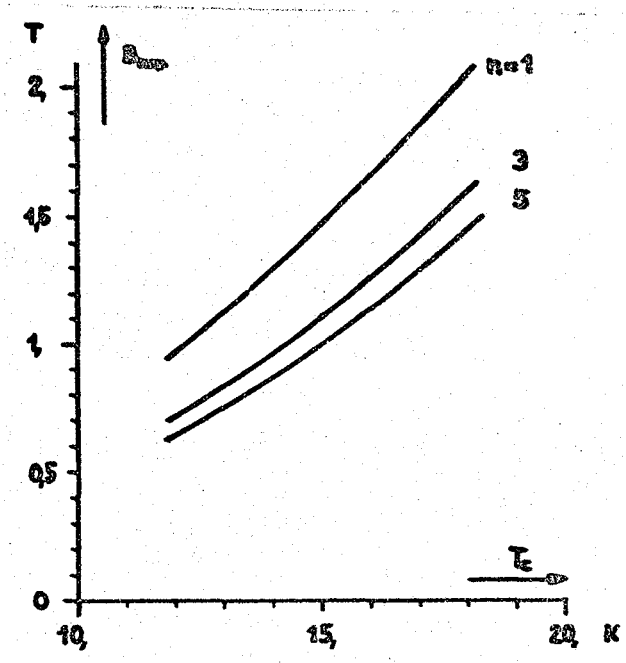

Figure 8

Maximum magnetic induction of the cycle as a function of the pre-cooling temperature. J-T cycle + magnetic cycle; 0. W* at 4.2° K.

n = number of double pairs of column (GGG)
Column dimensions: 15 cm long; cross-section warm end 0.5 cm^2; cross-section cold end 1.5 cm^2.

superconductor are based on existing technology. A critical current density of 1,000 A/mm^2 in the filament was assumed. Our calculations indicate that the maximum field on the conductor is on the order of 6T in the present case. The NbTi superconducting alloy requires 5.5-5.8° K while the Nb_3Sn alloy requires 7-8° K to achieve this level of performance.

In all cases an exterior ring of FeCo (saturation induction 2.4T; d = 8.3g/cm^3) was assumed. p.36

*Trans. note - Original says 0.W; it probably is 0.1W.

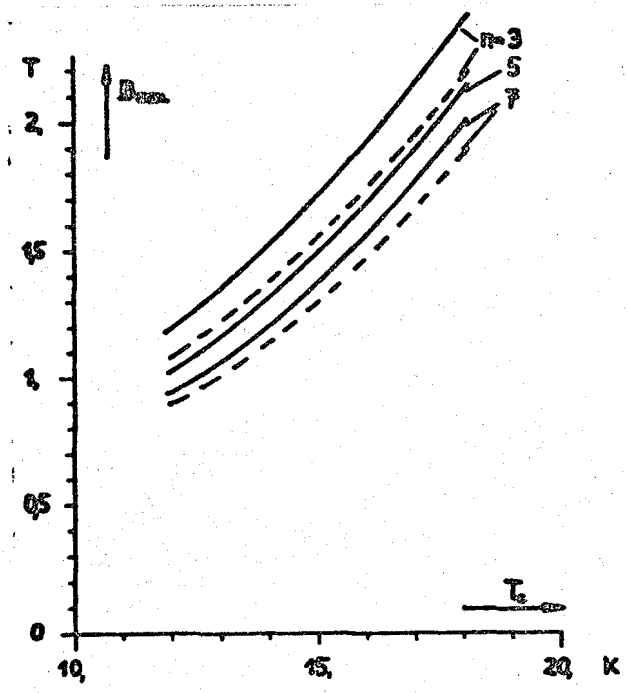

Figure 9

Maximum magnetic induction of the cycle as a function of the pre-cooling temperature. Magnetic cycle only. 0.1W at 4.2° K.

n = number of double pairs of columns (——GGG, - - -GGG + DAG)

Column dimensions: 15 cm long; cross-section warm end 0.5 cm^2; cross-section cold end 1.5 cm^2.

We sought to pre-dimension magnetic systems producing induction profiles of magnetic cycles whose maximum fields are shown in Figures 8 and 9. The main tool in this work was the POISSON numerical program which is well known to specialists (resolution of a two dimensional non-linear Laplace equation).

From the results obtained we arrived at the following conclusions:

Taking into consideration the geometry of the turning magnetic system and the form of the specified field profile, permanent magnets allow maximum values of the field to be around 1T.

Superconducting windings combined with polar pieces in FeCo allow the creation of field profiles in the desired form with $B_{max} \sim 2.5T$.

Superconducting windings have the advantage over permanent magnets in allowing the production of stronger magnetic fields. The encumbrances and mass of the two systems are essentially the same order of magnitude. Complex technical problems caused by the cooling required and the electric current needed (tens of amperes) for turning superconducting windings present serious inconveniences, particularly since it involves space applications. It should be remembered that the operating temperatures of superconductors are well below the pre-cooling temperatures envisioned. Consequently, this solution is hardly useful except in the J-T cycle/magnetic cycle design. A single magnetic stage with a superconducting winding could not start by itself!

In the framework of the rough draft which concerns us here there are two solutions which appear technically possible:

A single magnetic stage refrigerator design is compatible with a permanent magnet system. Such a choice implies a pre-cooling temperature of approximately 12^{o} K and the utilization of 5 double pairs of GGG columns in the annular air-gap (1.4 kg of GGG). A Nd-F-B based rotor has a diameter of approximately 22 cm and the total weight (rotor + stator + active material) is estimated to be 37 kg. p. 38

The refrigerator conceived as a J-T stage "doped" with a magnetic cycle can function with a magnetic system using a superconducting winding. A pre-cooling temperature of 18^{o} K and 1 double pair of columns of active material (270 g) is sufficient to attain the refrigeration power specified. The diameter of the entire magnetic system would be approximately 20 cm and the total weight (rotor + stator + active material) would be 36 kg (the external ring is 80%!)

The active magnetic material is assumed to be GGG in both versions. The use of a mixed GGG + DAG matrix does not lead to any significant gains particularly for low field cycles (cf. Figure 9). In the future more significant progress can come from ferromagnetic materials if it is possible to prepare

them with the desired composition and in a form which renders them utilizable for magnetic refrigeration [M.E. Wood, W.H. Potter, Cryogenics (1985) 25, 667].

5. COMPARATIVE ANALYSIS OF THE WEIGHT OF THE ENTIRE REFRIGERATION SYSTEM p.39

Equipment weight is a parameter of primary importance when it concerns space applications. It also appeared useful to us to establish a comparison of this subject among the three approaches envisioned for refrigeration at 4^{o} K; i.e., magnetic cycle alone, J-T cycle with an appended magnetic cycle, and J-T cycle alone.

First of all, one must recall that for pre-cooling at 10-20^{o} K it was concluded that the electric power absorbed by the refrigerator is constant and approximately 1,700W for the range of refrigeration energies required in our case (cf. Chapter 2.2). It is plausible to assume that the weight of this machine, in large part determined by the weight of the motors and the special bearings (space version!) is also constant and thus the same for all three designs described above. Its value is difficult to estimate in the absence of a definitive construction plan. One can, however, estimate that it will be approximately 60 kg. (cf. Final Report of an Evaluation Study of Mechanical Refrigerator Designs Applicable to Space Missions, Study Contract ESTEC no. 5629/83/NL/PB).

The actual weight of the 4^{o} K stage in the single magnetic cycle assumption comes primarily from the weight of its magnetic system (permanent magnets; cf. Chapter 4.2) which has been estimated at approximately 40 kg. Assuming that auxiliary elements such as bearings and magnetic rotor drives will add 10 kg, one arrives at a total weight of approximately 50 kg (not counting pre-cooling). The electric power necessary to keep the magnetic system moving (several watts) p.40 is considered negligible in its contribution to the weight we are interested in here.

If the approach to the 4^{o} K stage is a single J-T cycle the weight to be taken into consideration comes primarily from the compressor--estimated to weigh 10 kg. However, the electric power used by this compressor, estimated to be approximately 300W (cf. reference cited above, p. 41), includes an increase in the total weight load because of the weight associated with the equipment

needed to generate this power and for heat removal from the compressor. Relying on the figures given in the above cited document (p. 41-42), one can calculate that a 300W electrical feed system requires between 21 and 60 kg and a heat removal system requires between 5.8 and 6.1 kg. The lower numbers refer to a geo-stationary orbit while the higher numbers refer to a lower orbit. Adding up all of these items results in a total weight of 37 to 76 kg for a single J-T cycle (not counting pre-cooling).

Compared to the two previous approaches, the combination of a J-T stage with a magnetic cycle certainly represents the greatest weight. In effect, this design requires the use of both a magnetic system and a J-T compressor. It is true that the compressor in this case will weigh less than in a system using only a J-T cycle, principally because of the reduced compression rate (4 bar). Roughly estimating that this results in a weight reduction of all associated equipment by ½, one can calculate the total weight to be 18 or 38 kg depending upon which orbit is assumed. The entire assembly of this stage will thus weigh between 68 and 88 kg. p.41

Table I recapitulates the comparative analysis. The numbers indicate that on the average the single magnetic cycle and the single J-T cycle are equivalent as far as their weight is concerned and the J-T cycle + appended magnetic cycle involves the greatest weight. In all of these cases the pre-cooling system, with its associated equipment, represents between 80 and 85% of the total weight.

TABLE I

Comparative Analysis of the Weight of the Refrigerator

	Magnetic cycle only	J-T cycle only	J-T cycle + mag. cycle
Pre-cooling			
-Weight proper	60kg	60kg	60kg
-Elec. power	1500W	1500W	1500W
-Elec. feed	100-300kg	100-300kg	100-300kg
-Heat removal	40-50kg	40-50kg	40-50kg
SUB TOTAL	200-410kg	200-410kg	200-410kg
4^{o} K Stage			
-Weight proper	50kg	10kg	5 + 50kg
-Elec. power	<5W	300W	150W
-Elec. feed	<1kg	41*-60kg	10-30kg
-Heat removal	<1kg	6kg	3kg
SUB TOTAL	50kg	37-76kg	68-88kg
TOTAL WEIGHT	250-400**kg	237-486kg	268-498kg

6. CONCLUSIONS

The present study showed how magnetic refrigeration cycles could be used in a refrigerator with space applications and which could reach liquid helium temperatures. The elaboration of a rough draft allowed the definition of two versions for such a refrigerator. p.42

The first version includes the combination of a J-T cycle operating at low pressure (4 bar) and a magnetic cycle with a superconducting magnet. Although appearing very interesting from a thermodynamic stand point, this approach is less desirable because of its extreme technical complexity and

*Trans. note- Although the weight is 41 in the original, the sub-total and written description suggests that the correct number is 21.

**Probably 460.

and its weight.

In the second version the refrigeration occurs with only a magnetic cycle. The magnetic system is basically permanent magnets. The weight of the entire assembly is approximately equivalent to that of a classical J-T stage (15 bar) if one includes the weight of the equipment associated with the electrical feed and compressor cooling.

It appears premature at this time to establish a specific development plan for a magnetic refrigerator for space applications. In effect, magnetic refrigeration in the range of temperatures under consideration is still in the research stage and it is best to await the results of programs currently in progress which aim to experimentally demonstrate the envisioned processes. p.43

There is, however, a line of development which can be dealt with immediately--the pre-cooling system. Heat removal of magnetic refrigerators of the type evaluated in this study is compatible with the capacities of mechanical "breeder" refrigerators. These systems are manufactured industrially for terretrial applications. However technologies dealing with the constraints of long lasting space missions do not yet exist and thus must be developed. In addition, this development is essentially the same regardless of the choice of the cold stage;i.e., regardless of whether it is a magnetic cycle or a classical J-T cycle. In effect, the respective levels of pre-cooling power are in a range in which are found many "breeder" machines. In particular, their electrical consumption is practically independent of their useful capacity.

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